FLOW CONTROL SYSTEM FOR A GAS TURBINE ENGINE

BACKGROUND OF THE INVENTION

1. Field of the Invention

[0001] The subject invention is directed generally to a system for regulating fluid flow, and more particularly, to a system for regulating the flow of liquid fuel from a variable displacement pump to a gas turbine by utilizing bypass flow.

2. Background of the Related Art

[0002] Fixed delivery fuel pumps have often been over-sized to provide excessive fuel flow capacity in order to insure adequate supply to the associated engine. Consequently, under many operating conditions, large amounts of pressurized fuel are returned to the pump inlet for recirculation. The return and recirculation results in significant fuel heating due to additional energy being put into the fuel which is subsequently turned into heat as the pressure drops in the recirculation path. In modern designs, fuel heating is a critical issue because the fuel is typically used as a heat exchanger to maintain proper operating temperature. Other methods of heat exchange are undesirable because of the associated size, weight and cost. Such concerns are magnified in modern engines because the fuel pumps also need to supply fuel to engine geometries. For example, modern mid to large class engines utilize linear pistons as guide vanes. The linear pistons require a significant source of fuel to slew. This slewing is a transient event that can unacceptably starve the supply of fuel to the engine.

[0003] Variable displacement fuel pumps have partially overcome the drawbacks of fixed delivery pumps by being able to vary the amount of fuel output. By varying the fuel output, the fuel delivered more closely matches engine demand. Thus, the recirculated flow, along with the heat generated thereby, is reduced. Variable displacement fuel pumps are known in the art as disclosed in U.S. Patent No. 5,833,438 to Sunberg, the disclosure of which is herein incorporated by reference in its entirety. A variable displacement pump typically includes a rotor having a fixed axis and pivoting cam ring. Controlling the position of the cam ring with respect to the rotor controls the output of the pump. The output flow may be controlled by a torque motor operated servo valve. However, the engine operating conditions often include transients such as those caused by engine actuator slewing, start-up and the like as would be appreciated by those of ordinary skill in the pertinent art. Under such rapidly varying operating conditions, prior art pump control systems have been unable to respond quickly and adequately. Moreover, many prior art pump control systems lack the required stability to reliably provide fuel to the engine. So despite the advances of the state of the art, variable displacement pumps are lacking in stability and still do not respond quickly enough to varying engine demands. As a result, poor performance and excess fuel flow are still common.

[0004] Examples of variable displacement pump control arrangements are disclosed in U.S. Patent Nos. 5,716,201 to Peck et al. and 5,715,674 to Reuter et al., the disclosures of which are herein incorporated by reference in their entirety. These pump control systems attempt to maintain accurate fuel flow throughout the range of engine operating conditions. However, as noted above, such systems still contain inadequacies such as complexity. Moreover, such

systems can only achieve adequate bandwidth by delivering excessive fuel which must be recirculated. It is also undesirable for pump control systems to include sophisticated electronics and numerous additional components that undesirably increase costs and complexity.

[0005] In view of the above, it would be desirable to provide a flow control system which has a simple design for quickly regulating the output flow of a variable displacement pump with stability and without the associated drawbacks of the prior art.

SUMMARY OF THE INVENTION

[0006] In one embodiment, the subject invention is directed to a flow control system for controlling a variable displacement pump including a metering valve in fluid communication with the pump for metering an output of the pump. A regulating valve for maintaining a pressure differential across the metering valve receives a portion of the output of the pump as a bypass flow at a first pressure, wherein an output of the regulating valve is at an interim pressure, wherein the interim pressure is equal to an approximate average of the first pressure and a low reference pressure. An actuator sets a displacement of the pump by acting on a piston connected to a cam ring of the pump. The setting of the actuator is determined by a differential between the interim pressure and a second portion of the output of the pump at the first pressure.

[0007] It is an object of the present disclosure to increase the fuel metering unit response while maintaining acceptable stability at all operating conditions.

[0008] It is another object to provide a hydromechanical fuel metering unit for a variable displacement pump. It is still another object to provide a fuel metering unit that achieves quick and accurate response to dynamic flow conditions.

[0009] In another embodiment, the subject invention is directed to a method for metering a variable displacement pump that provides fuel to an engine, the method includes the steps of receiving fuel at a low reference pressure into the variable displacement pump, pumping the fuel through the pump such that an output of the pump is at an elevated pressure, metering the output of the variable displacement pump with a metering valve, creating a spill return flow from the output of the variable displacement pump to allow for quick response when additional fuel is required by the engine, regulating a pressure differential across the metering valve with a regulating valve. The regulating valve is in fluid communication with the spill return flow to generate an interim pressure substantially equal to an average of the spill return and the low reference pressure. The method also includes the step of adjusting a displacement of the pump with a cam actuator connected to a cam ring of the variable displacement pump for adjusting the output, wherein the cam actuator receives the interim pressure to determine a setting of the cam actuator.

BRIEF DESCRIPTION OF THE DRAWINGS

[0010] So that those having ordinary skill in the art to which the subject invention appertains will more readily understand how to make and use the same, reference may be had to the Sole Figure wherein:

[0011] The Sole Figure is a schematic representation of a flow control system constructed in accordance with the subject invention.

DETAILED DESCRIPTION OF PREFERRED EMBDOMENTS

[0012] Referring now to the Sole Figure, there is illustrated a schematic representation of a flow control system in accordance with the subject invention which is designated generally by reference numeral 10. For clarity throughout the following description, arrows are shown within the lines of system 10 to indicate the direction in which the fuel flows and an annotated letter "P" is shown to indicate a pressure at certain locations. All relative descriptions herein such as left, right, up, and down are with reference to the system 10 as shown in the Sole Figure and not meant in a limiting sense. Additionally, for clarity common items such as filters and shut off solenoids have not been included in the Sole Figure. The system 10 maintains the output flow of a variable vane displacement pump 12 to provide fast response to engine needs in a stable manner yet excessive complexity is avoided.

[0013] The pump 12 includes a rotor 14 and a pivoting cam ring 16. For a detailed description of a variable displacement vane pump, see U.S. Patent Application Publication No. 2002/0103849 published on June 5, 2003 which is incorporated herein by reference in its entirety. The pump 12 receives fuel flow at an inlet pressure P_{AF}, and delivers fuel flow at an output pressure P_F. A piston 18 is operatively connected to the cam ring 16 to control the position of the cam ring 16 relative to the rotor 14 and, thereby, vary the output flow of the pump 12. A cam actuator assembly 20 positions the piston 18 as described below. It should be appreciated by those of ordinary skill in the art that other types of actuators similarly and differently arranged would perform this same function and are, therefore, considered mere design

choices well within the scope of the subject invention as claimed. The maximum flow setting of the pump 12 occurs when the piston 18 is moved the maximum distance to the left.

[0014] A feedback line 21 in fluid communication with the output of the pump 12 provides fuel at pressure P_{1W} to a line 29 connected to an inlet 22 of the cam actuator 20. Orifices 24 and 26 limit the flow into line 29. The pressure in line 29 is approximately equal to $P_{1W} + P_{AF}$ divided by two, and designated as P_{I1} in the Sole Figure. It will be appreciated by those of ordinary skill in the art that the pressure at P_{1W} will be substantially equal to the pressure P_F. The feedback line 21 also provides fuel at pressure P_{1W} to other locations not shown such as the engine geometry, main metering valve and bleed band servos (not shown) as required. Line 29 also connects to low reference pressure P_{AF}. Another inlet 28 of the cam actuator 20 receives fuel at an interim pressure P_{I2} as will be described hereinbelow.

[0015] A housing 23 of the cam actuator 20 retains the piston 18 for dividing the interior of the housing 23. A coiled spring 30 biases the piston 18. Within the housing 23, the pressure on the right side of the piston 18 is approximately equal to the average of P_{1W} and P_{AF}. The combination of the pressure differential between the right and left sides of the housing 23 together with the sizing of a spring 30 act to position the piston 18 within the cam actuator 20. The cam ring 16 moves correspondingly and the output of the pump 12 varies. Preferably, the spring 30 is sized and configured to position the piston 18 at maximum flow for start-up of the pump 12. Throughout system 10, springs are sized as a function of the product of piston area and fuel pressure as would be appreciated by those of ordinary skill in the art and therefore not further described herein.

[0016] The output of the pump 12 passes through a wash filter 32 for cleaning debris prior to entering a main metering valve 34 and line 21. The main metering valve 34 is disposed between the pump 12 and engine (not shown) for providing fuel to the engine at a selected rate and pressure P_M. The main metering valve 34 insures that P_F is greater than P_M by some pre-selected, substantially constant value. Suitable main metering valves 34 are well known in the prior art and therefore not further described herein. The preferred metering valve 34 performs the function of selectively varying the amount of fuel passing therethrough. The main metering valve 34 receives fuel at pressure P_F and the fuel exits at pressure P_M.

[0017] A line 35 connects the output of the pump 12 to a bypassing pressure regulator valve assembly 36. The flow in line 35 is referred to as the spill return flow at Pressure P_F. The regulator valve assembly 36 includes a housing 38 defining an interior with a spring-biased spool 40 operatively disposed therein. A left face of the spool 40 has fuel at pressure P_F there against. A metering head adjustment screw 42 is attached to the spool 40 for calibrating the position of the spool 40 within the regulator valve assembly 36 during set up. The housing 38 defines an inlet 44 connected to line 35 for receiving fuel at pressure P_F. Another inlet 46 of the housing 38 receives fuel at pressure P_M from static flow line 37 to dampen the motion of the spool 40. An orifice 48 is disposed in the line 37 for dampening. The housing 38 also defines a restricting outlet 50 for the fuel to exit from the regulator valve assembly 36. The outlet 50 is in fluid communication with the inlet 28 of the pump 12 via line 51. Line 51 also connects the low reference pressure P_{AF}, wherein an orifice 52 limits the flow to the low reference pressure P_{AF}. The pressure within line 51 is approximately equal to the average of P_F and P_{AF}, hereinafter

designated the interim pressure P_{I2}. The combination of the pressure differential between P_F on the left side of the housing 40 and P_M on the right plus the spring-biasing of the spool 40 ultimately positions the spool 40.

[0018] During steady-state operation, the left side of regulator valve assembly 36 and the right side of the cam actuator 20 are at approximately the same pressure. The recirculation flow through the regulator valve assembly 36 is maintained at a low level by the spool 40 partially blocking the outlet 50 yet the system 10 can rapidly and sufficiently respond to transient events. In the preferred embodiment, the spool 40 within the regulator valve assembly 36 is maintained substantially at a nominal position during normal operation. In most modern engines, the variation in fuel demand as a result of running the engine is relatively minor compared to that associated with the slewing of engine geometries.

[0019] When a transient event occurs, the pump 12 must produce more fuel rapidly. For example, a transient event would be an actuator motion by utilization of the slewing source at pressure P_{1W} in line 21. Increased demand by the transient event causes the main metering valve 34 to respond by opening to immediately increase flow to the engine and starts a chain of events which leads to an increase in the output of the pump 12. The pump 12 cannot immediately respond with increased displacement so the incremental demand comes from diminished spill return flow in line 35. As a result, the pressure P_M increases and the Pressure P_F decreases (i.e., a drop of the pressure differential (P_F-P_M) across the main metering valve 34). The regulator valve assembly 36 senses the pressure differential drop and the spool 40 strokes to the left. The pressure in the outlet 50 is decreased and, in turn, the pressure into the left side of the cam

actuator 20 drops. The decrease in pressure of the left side of the cam actuator 20 causes the piston 18 to stroke to the left. When the piston 18 strokes to the left, the output of the pump 12 increases. The increased pump output raises the pressure P_F until the pressure differential across the main metering valve 34 returns to the nominal steady-state level and the steady-state condition is reattained.

[0020] In the alternative, when a transient event occurs where the pump 12 must rapidly decrease the output to prevent excessive recirculation, the main metering valve 20 responds by closing to decrease flow to the engine and starts a chain of events which leads to a decrease in the output of the pump 12. The pump 12 cannot immediately respond with decreased displacement so the decreased demand results in a decrease of pressure P_M and an increase in Pressure P_F (i.e., a rise of the pressure differential (P_F-P_M) across the main metering valve 34) to allow the system 10 to immediately respond. The regulator valve assembly 36 senses the pressure differential rise and the spool 40 strokes to the right. As a result, the pressure in the outlet 50 is increased and, in turn, the pressure into the left side of the cam actuator 20 rises. The rise in pressure of the left side of the cam actuator 20 causes the piston 18 to stroke to the right. When the piston 18 strokes to the right, the output of the pump 12 decreases until the pressure differential (P_F-P_M) across the main metering valve 34 returns to the nominal steady-state level with the spool 40 substantially at a nominal position with the regulator valve assembly 36.

[0021] In summary, the regulator valve assembly 36 is used to minimize the recirculation flow, while allowing the system 10 to respond quickly to transient demands. The recirculation flow is regulated and the position of the spool 40 is at a substantially nominal position during

steady-state operation. Pressures substantially equal to the average of the low reference pressure and output pressure of the pump are utilized to set the regulator valve and cam actuator.

[0022] While the subject invention has been described with respect to preferred embodiments, those skilled in the art will readily appreciate that various changes and/or modifications can be made to the invention without departing from the spirit or scope of the invention as defined by the appended claims.